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TITLE: A THEORETICAL PREDICTION OF CRITICAL HEAT FLUX IN SATURATED POOL BOILING DURING POWER TRANSIENTS

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NOMENCLATURE

A_v	Heater area covered by vapor (m^2)
A_w	Total heater area (m^2)
B_s	Switch-over parameter (dimensionless)
c	Wave velocity at the interface (m/s)
C_p	Specific heat of the heater material (J/kg)
d	Heater diameter (m)
d_e	Equivalent diameter (m)
$f_1(P), f_2(P)$	Prescribed functions of pressure
g	Gravitational constant (m/s^2)
$H(\ )$	Heavyside step function
h_c	Convective heat-transfer coefficient (W/m^2-K)
h_{fg}	Latent heat of vaporization (J/kg)
k	Conductivity of the heater material (W/m-K)
P	Pressure (MPa)
Q	Power generation rate (W/m^3)
q	Surface heat flux (W/m^2)
q_b	Heat flux at switch-over point (W/m^2)
$q_{CHF,SS,OO}$	Steady-state CHF in saturated pool boiling (W/m^2)
$q_{CHF,TR}$	Transient CHF (W/m^2)
R	Heater radius (m)
r	Radial coordinate (m)
S	Total surface area of the heater (m^2)
T	Temperature (K)
t	Time (s)
$t_{CHF,SS}$	Time the transient surface heat flux reaches steady-state CHF (s)
$t_{CHF,TR}$	Time the transient surface heat flux reaches transient CHF (s)
u	Velocity (m/s)
V	Total volume of the heater (m^3)
v_1	Volumetric growth rate of a bubble (m^3/s)

W	Width of the flat ribbon heater (m)
δ	Liquid layer thickness (m)
δ_c	Critical liquid-layer thickness (m)
$\delta_{c,o}$	Critical liquid-layer thickness in saturated pool boiling at steady-state CHF level (m)
δt	Small time increment (s)
λ	Interfacial disturbance wavelength (m)
λ_H	Helmholtz unstable wavelength (m)
λ'_{T}	Modified Taylor unstable wavelength (m)
ξ	The volumetric ratio of the accompanying liquid to the moving bubble (dimensionless)
η	Ratio of transient CHF to steady-state CHF (dimensionless)
σ	Surface tension (N/m)
ρ	Density (kg/m^3)
τ	Exponential period (s)
τ_d	Hovering period (s)

Subscripts:

f	Saturated liquid
g	Saturated vapor
i	Initial value
w	Wall condition

A THEORETICAL PREDICTION OF CRITICAL HEAT FLUX IN SATURATED POOL BOILING DURING POWER TRANSIENTS*

by

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ABSTRACT

Understanding and predicting critical heat flux (CHF) behavior during steady-state and transient conditions is of fundamental interest in the design, operation, and safety of boiling and two-phase flow devices. Presented within this paper are the results of a comprehensive theoretical study specifically conducted to model transient CHF behavior in saturated pool boiling. Thermal energy conduction within a heating element and its influence on the CHF are also discussed. The resultant theory provides new insight into the basic physics of the CHF phenomenon and indicates favorable agreement with the experimental data from cylindrical heaters with small radii. However, the flat-ribbon heater data compared poorly with the present theory, although the general trend was predicted. Finally, various factors that affect the discrepency between the data and the theory are listed.

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** Work performed while a graduate student at the University of Central Florida, Orlando, Florida.

I. INTRODUCTION

Boiling heat transfer with time-dependent heat input, as well as the prediction of the critical heat flux (CHF) under such conditions, is of interest in several nuclear reactor safety applications. One application in light-water nuclear reactor technology involves the reactivity-initiated accident (RIA), in which a sudden increase in power generation may occur. Therefore, an accurate modeling of CHF is required to evaluate RIA scenarios. Consequently, several investigators [1–10] have concerned themselves with related studies.

Along with power-burst experiments to simulate an RIA within reactor environments, [1–3] there have been several studies toward providing a fundamental understanding of the CHF phenomenon [4–10]. Tachibana *et al.* [4], Sakurai *et al.* [5], and Kawamura *et al.* [6] studied the transient boiling experimentally using electrically heated flat-ribbon heaters. These experiments were conducted under atmosheric pressure. The effect of subcooling on the transient boiling is also investigated in the experiments of Sakurai, *et al.* [5] and Kawamura *et al.* [6]. Sakurai *et al.* [5] were able to empirically correlate their saturated and subcooled boiling data as a function of the exponential period of the power transient. The transient pool boiling with cylindrical horizontal wire of small radius has also been studied experimentally. The studies of Sakurai and Shiotsu [7, 8] and Kuroda (as cited by Serizawa [9]) are such examples. In the studies of Sakurai and Shiotsu [7, 8], saturated pool boiling at different pressures, ranging from atmospheric to about 2 MPa, is investigated. An empirical correlation for the transient CHF is also obtained [8]. In addition, Kuroda (as cited by Serizawa [9]) also obtained experimental CHF data in transient subcooled pool boiling. The experimental setup of Kataoka *et al.* [10] consisted of a vertical wire of small radius placed in a flow channel. It was basically designed to study the transient boiling in forced convective conditions. However, a set of data for natural convection boiling is also obtained. Serizawa [9] studied the transient CHF theoretically. He was able to correlate the transient CHF data under various conditions using a single CHF model. However, such a model does not include all the governing physics of the problem, and the final correlation requires extensive empiricism.

The present authors focused on theoretically modeling CHF behavior during power transients in saturated and subcooled pool boiling and forced convective boiling. This paper contains the first part of this study, which models the transient CHF in saturated pool boiling. Theoretical models for subcooled pool boiling and forced convective boiling will be published subsequently. The model of Haramura and Katto [11] is used as the steady-state CHF model in saturated pool boiling and is summarized in the following section. A transient CHF model is developed in Sec. III. Section IV discusses the effect of the transient conduction within the heater on the CHF model developed in the previous section. In Sec. V, the correlations resulting from the present theory are compared with the transient CHF data available in the literature. Finally, Sec. VI summarizes and concludes the present paper.

II. CHF IN STEADY-STATE SATURATED POOL BOILING

A theoretical prediction of steady state CHF in saturated pool boiling was first presented in the foundation work of Zuber [12]. His analysis, however, was subsequently questioned because of its limited modeling of the governing physics [13]. Zuber's model [12] suggests that the collapse of the vapor removal mechanism is solely responsible for CHF. In 1983, Haramura and Katto [11] presented what is perhaps the most complete formulation of saturated steady state pool boiling CHF. This model, based on experimental observations, is referred to as the

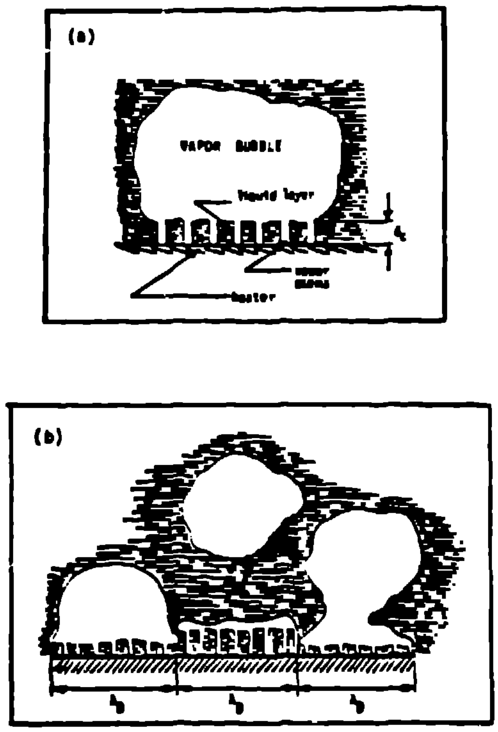

Fig. 1.
Boiling configuration at high heat fluxes over flat plates [11].

"multistep" model [14] (see Fig. 1). As shown in Fig. 1, high heat fluxes are postulated to cover the surface by numerous bubbles, and the liquid supply to the liquid layer from the bulk is blocked. The only time the bulk liquid reaches the surface is when the bubbles depart, and then the local initiation of a new bubble is almost instantaneous. The hovering period of the bubble, τ_d, may be estimated by solving the equation of motion of the idealized bubble [11]. The critical liquid-layer thickness is estimated [11] to be a fraction between 0 and 0.5 of the Helmholtz unstable wavelength in the vapor stems underneath the hovering bubbles. This fraction is assumed as 0.25 by Haramura and Katto [11]; thus,

$$\delta_c = 0.25\lambda_H = f_1(P)/q^2 \quad . \tag{1}$$

where

$$f_1(P) = 0.5\pi\sigma[(\rho_f + \rho_g)/(\rho_f\rho_g)](A_v/A_w)^2(\rho_g h_{fg})^2 \quad . \tag{2}$$

Haramura and Katto [11] postulate that CHF will occur if the heat flux is high enough to evaporate the critical liquid layer $\delta_{c,o}$ prior to the end of the hovering period. This is mathematically expressed by the following equation:

$$\tau_d \, q_{CHF,SS,0} = \rho_f \delta_{c,o}(1 - A_v/A_w) h_{fg} \quad . \tag{3}$$

where A_v/A_w represents the ratio of the surface area covered by vapor to the total heated area.

In the following section, a transient CHF model is developed based upon the steady-state model of Haramura and Katto [11]. The final results of the general mathematical model are obtained for exponential and linear power transients.

III. TRANSIENT CHF MODEL

Although the model of Haramura and Katto [13] was developed for steady-state conditions, it is also useful in the understanding of the transient CHF. The correlation, given by Eq. (3), is explicitly written in terms of a time constant for the CHF phenomenon, which is the hovering period τ_d. This suggests that once the steady-state CHF is applied to the heater surface, dryout can be detected after a period of time, which is τ_d. During transient power conditions, the local heat flux increases during the hovering period, and the complete evaporation of the liquid layer takes place sooner. By the time the surface is essentially dry, the local heat flux reaches a value higher than at steadystate. This transient heat flux is referred to as $q_{CHF,TR}$.

During the bubble-hovering period, no liquid is supplied to the liquid layer; thus, the rate of thinning of the liquid layer is governed by one of two mechanisms: hydrodynamic instability (which dictates the maximum thickness of the stable liquid layer for a given surface heat flux) or evaporation. This may mathematically be expressed as

$$(d\delta_c/dt) = MAX\ \{(\partial\delta_c/\partial q)(dq/dt),\ q/f_2(P)\}\quad . \tag{4}$$

where

$$f_2(P) = \rho_f h_{fg}\ (1 - A_v/A_w)\quad . \tag{5}$$

Previously, the thinning rate of the liquid layer was mistakenly postulated to be the sum of the two mechanisms [15]. However, later considerations suggested the more accurate form given by Eq. (4).

The first term on the right hand side (RHS) of Eq. (4) corresponds to hydrodynamic stability behavior and, for steady state conditions, is given by Eq. (2). If it is assumed that the velocity in the vapor stems accelerates instantaneously to a value dictated by the surface heat flux then Eq. (2) can likewise approximate the transient behavior. This assumption is considered reasonable since the length of the vapor stems (the distance traveled by the generated vapor before it reaches the bubble) is very small (on the order of 10 to 100 microns). A numerical justification of this assumption is given in Appendix B. The hypothesis formulated by Eq. (4) is shown in Fig. 2. This figure shows that if the heat flux is kept constant at the steady-state CHF level, the liquid layer will evaporate at the end of the hovering period τ_d. But, because the heat flux increases with time, the rate of evaporation also increases, as shown by the dotted lines. However, an increase in the heat flux also causes a decrease in the thickness of the stable liquid film, as shown by the solid line. The slope of the evaporation line and critical liquid thickness line determine the dominant mechanism. Figure 2 shows that between points A and B, the hydrodynamic instability is dominant, whereas at point B the evaporation starts dominating until the complete evaporation of the liquid layer thickness. Therefore, the liquid layer thickness follows the curve ABC rather than AC', as suggested by Serizawa [9].

The hypothesis illustrated in Fig. 2 also requires that the bubble remain on the surface between $t_{CHF,SS}$ and $t_{CHF,TR}$, which prevents any liquid supply from the bulk during this period. To satisfy this condition, the bubble must have a hovering period greater than the period from A to C. By solving the equation of motion of an idealized bubble, Pasamehmetoglu and Nelson [16] were able to show that the above requirement is satisfied during transient

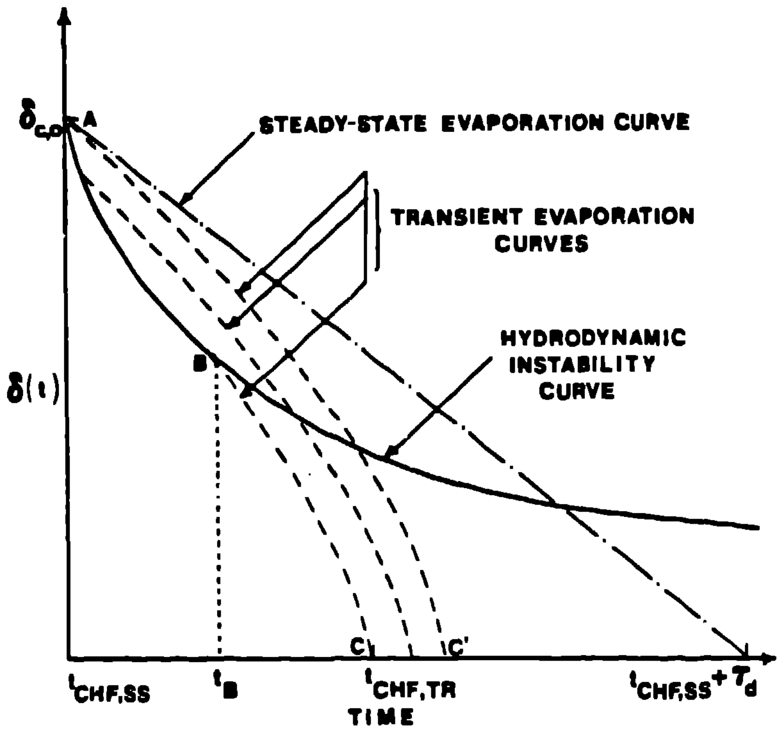

Fig. 2.
The change in liquid-layer thickness as a result of hydrodynamic instability and evaporation.

boiling. In this liquid layer thinning model, the switchover point, B, between the two mechanisms may be calculated by setting the slopes of evaporation and hydrodynamic instability lines equal, such that:

$$(\partial\delta_c/\partial q)\,(dq/dt) = \quad q/f_2(P) \quad . \tag{6}$$

Using Eq. (2), which relates the critical liquid-layer thickness to the surface heat flux, the heat flux at the switchover point may be calculated as

$$q_B = [2f_1(P)f_2(P)(dq/dt)_{q=q_B}]^{\frac{1}{4}} \quad , \tag{7}$$

which may be written in the following form:

$$q_B = B_s q_{CHF,SS,OO} \quad . \tag{8}$$

In Eq. (8), B_s is a parameter that determines the switchover between the two mechanisms and is obtained by using Eqs. (1) and (3) as follows:

$$B_s \quad [2\tau_d(dq/dt)_{q=q_B}/q_{CHF,SS,OO}]^{\frac{1}{4}} \quad . \tag{9}$$

If B_s is less than or equal to 1, the decrease in the liquid layer is controlled by evaporation alone. However, if B_s is greater than 1, the effect of the hydrodynamics on the decrease of liquid layer thickness must be included in the analysis. Once the surface heat flux is known as a function of time, the switchover heat flux, q_B, may be calculated through Eqs. (8) and (9). Hence, the transient CHF may be predicted through the following integrals:

$$(\delta_{c,o}) = \int_{t_{CHF,SS}}^{t_I} [2 f_1(P)(dq/dt)/q^3] dt + \int_{t_I}^{t_{CHF,TR}} [q/f_2(P)] dt, \quad \text{if } B_s > 1 \quad , \tag{10}$$

and

$$(\delta_{c,o}) = \int_{t_{CHF,SS}}^{t_{CHF,TR}} [q/f_2(P)] dt, \quad \text{if } B_s < 1 \quad . \tag{11}$$

In Eqs. (10) and (11), $\delta_{c,o}$ is the maximum liquid layer thickness at steady-state CHF level dictated by Helmholtz instability. It is given by Eq. (1), in which the heat flux, q, is replaced by the steady-state CHF, $q_{CHF,SS,OO}$.

If the surface heat flux is known as a function of time, Eqs. (10) and (11) may be integrated. For example, if the surface heat flux increases linearly, it may mathematically be expressed as:

$$q(t) = q_i + (dq/dt)t \quad . \tag{12}$$

For such a linear increase, it can be shown [17] that Eqs. (10) and (11) may be integrated to yield

$$\eta = [1 - H(B_s - 1)](1 - B_s^4)^{\frac{1}{2}} + [H(B_s - 1)](\sqrt{2}B_s^2) \quad , \tag{13}$$

where

$$\eta = q_{CHF,TR}/q_{CHF,SS,OO} \quad , \tag{14}$$

$H(B_s - 1)$ is the Heavyside step function and B_s is the switch-over parameter given by Eq. (9).

Similarly, if the surface heat flux increases exponentially according to

$$q(t) = q_i exp\,(t/\tau) \quad . \tag{15}$$

Eqs. (10) and (11) may be integrated to yield [17]

$$\eta = [1 - H(B_s - 1)](1 + \tau_d/\tau) + [H(B_s - 1)]1.89(\tau_d/\tau)^{\frac{1}{2}} \quad , \tag{16}$$

where the switch-over parameter B_s becomes

$$B_s = (2\tau_d/\tau)^{\frac{1}{2}} \quad . \tag{17}$$

In Eqs. (13) and (16), the Heavyside step functions identify the different thinning mechanisms for the liquid layer. In exponential transients, for instance, the existence of each mechanism depends upon the magnitudes of the hovering period, τ_d, and the exponential period, τ. As shown in Fig. 3, a mechanism map may be obtained by plotting Eq. (17). The boundary of the two mechanisms shown in Fig. 3 is obtained by setting B_s equal to unity in Eq. (17).

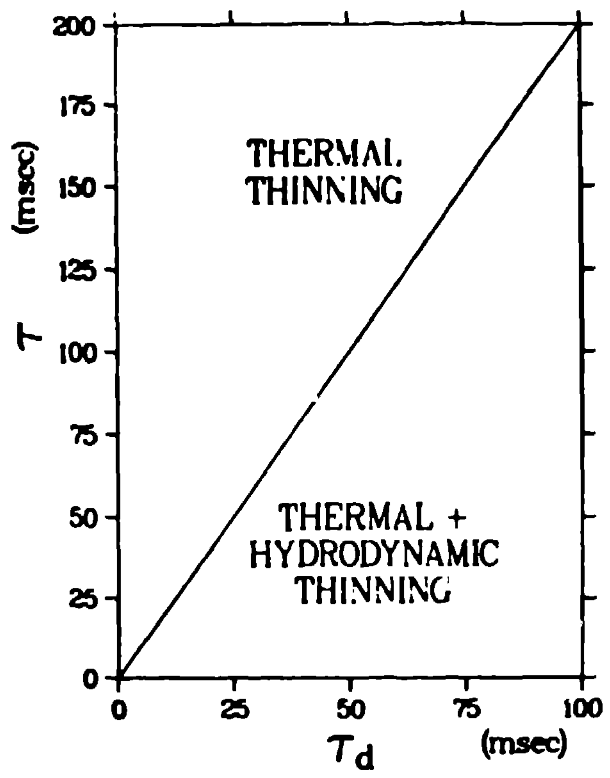

Fig. 3.
The mechanism map for liquid-layer thinning.

The use of Eqs. (13) and (16) further requires the quantitative knowledge of the steady-state hovering period, τ_d. This may be analytically obtained by solving the equation of motion of an idealized bubble, as done by Haramura and Katto [11]. The resulting correlation for τ_d is given in Appendix A. A plot of this correlation at steady-state CHF level as a function of the system pressure and the heater geometry is shown in Fig. 4. The correlations given by Haramura and Katto are used to estimate the CHF needed to obtain these plots. Figure 4 shows that for heaters of small diameter, the hovering period remains almost constant over a wide range of the system pressure. This result is expected because the experimental data of Sakurai and Shiotsu [8] show that the CHF ratio, η, is almost independent of the system pressure over a pressure range between atmospheric and 2 MPa.

IV. CONDUCTION WITHIN THE HEATER

The transient CHF correlations developed in the previous section are based on the rate of change of the local heat flux. During power transients, however, the rate of change of the heat-generation rate is often the controlled parameter, and it is usually directly measured or calculated. Therefore, in order to use the above CHF correlations, a relation between the heat-generation rate and the surface heat flux must be established. In this section, the transient conduction problem of a cylindrical heater in a pool of liquid is considered. The heat-generation rate is assumed to increase exponentially according to:

$$Q(t) = Q_i \exp(t/\tau) \quad . \tag{18}$$

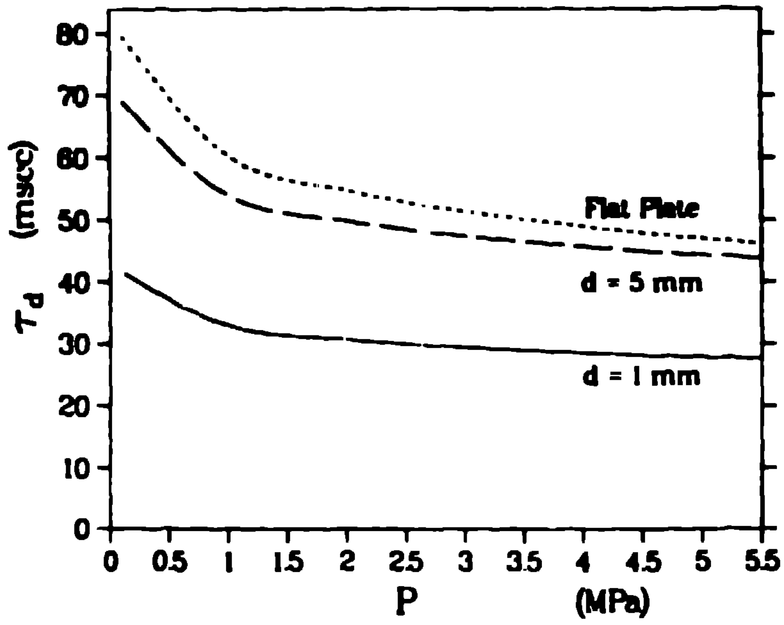

Fig. 4.
The bubble-hovering period as a function of heater geometry and pressure.

Neglecting the temperature variations in the axial and circumferential directions, the energy equation becomes:

$$\rho C_p \frac{\partial T}{\partial t} = k\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)\right] + Q_1 \exp\,(t/\tau) \tag{19}$$

subject to the following boundary conditions:

$$\text{at } r = 0, \quad \frac{\partial T}{\partial r} = 0 \ , \tag{20}$$

$$\text{and, at } r = R, \quad T = T_w \quad (\text{or, } \frac{\partial T}{\partial r} = h_c(T_w - T_{sat})). \tag{21}$$

The difficulty in obtaining an exact solution to Eq. (19) arises because of the boundary condition given by Eq. (21). In this equation, both T_w and h_c are functions of time and must be supplied from the solution of the conjugate convection problem. It can be shown [17] that the surface heat flux may be approximated through a simple relation as

$$q(t) = \frac{V}{S} Q(t) \ . \tag{22}$$

if the following conditions are satisfied: the heater has (1) a high thermal conductivity k, (2) low heat capacity ρC_p, and (3) small radius R, (4) the surface convective heat-transfer

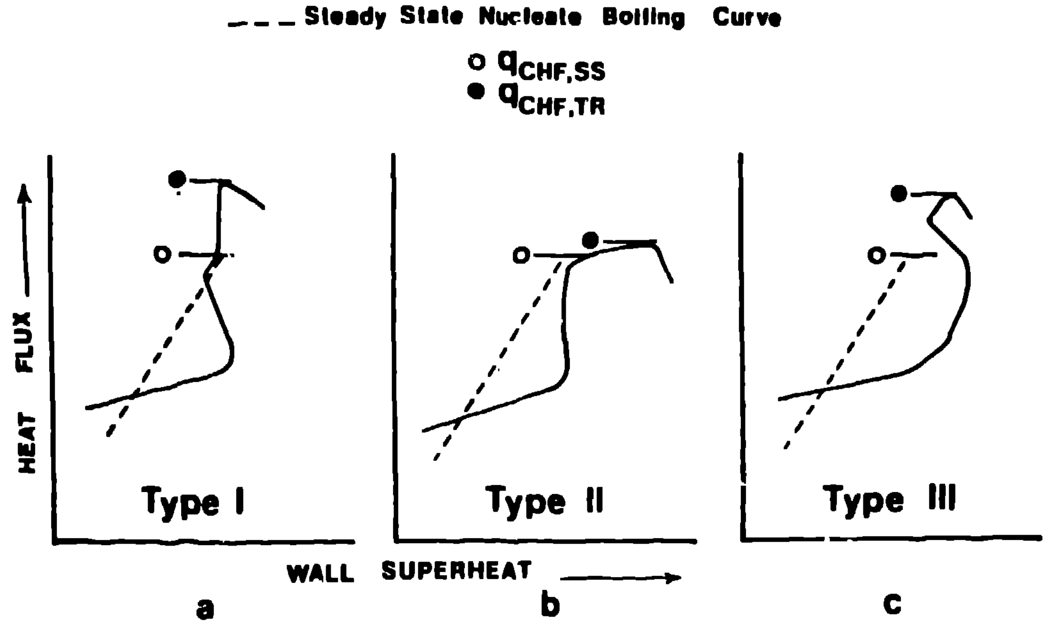

Fig. 5.
The different types of boiling during power transients.

coefficient h_c is high, and (5) the exponential period τ is not very small. This relation is referred to as quasi-steady conduction model. For electrical heaters, Sakurai and Shiotsu [7, 8] and Serizawa [9] experimentally showed that Eq. (22) may accurately predict the surface heat flux between $t_{CHF,SS}$ and $t_{CHF,TR}$ in many instances. However, the boiling curves obtained in the same studies suggest that two other conduction models may also be observed under certain conditions. The different boiling curves dictated by various conduction models are shown in Fig. 5. Figure 5a corresponds to quasi-steady conduction model given by Eq. (22). The boiling type of Fig. 5b is observed if the convective heat-transfer coefficient is not high enough, in which case most of the generated heat may not be convected away and remains stored in the heater. Under these conditions, the quasi-steady conduction model will overpredict the surface heat flux. On the other hand, if a sudden increase in h_c occurs (for instance, right after the initiation of boiling), there may not be enough heat generated and conducted to the surface to compensate for the suddenly augmented convection. Thus, some of the previously stored energy starts being convected away, which results in a decrease of the surface temperature as shown by Fig. 5c. Under these conditions, the quasi-steady conduction model will underpredict the surface heat flux. Hereafter, the boiling patterns shown in Fig. 5a, 5b, and 5c are referred to as *types I, II,* and *III*, respectively. Since *type-I* boiling is experimentally observed more often than the other two, the transient CHF correlations are coupled with the quasi-steady conduction model for comparison with the data.

V. COMPARISON WITH DATA

In this section, the present theory is compared to data from the experiments of Sakurai and Shiotsu [8], Kuroda (as cited by Serizawa [9]), Tachibana *et al.* [4], Sakurai *et al.* [5] and

Kataoka *et al.* [10]. Equations (13) and (16) are used to calculate the CHF ratio, η. To obtain the corresponding transient CHF, the computed values of η are multiplied by the steady-state CHF measured in the experiments. Such an approach is preferred because the primary purpose of the present study is not to evaluate the steady-state CHF correlations, but to develop a transient CHF model. In the experiments with cylindrical heaters, the measured steady-state CHF values were very close to the values predicted by the available correlation [11, 18]. However, in the experiments of Tachibana *et al.* [6] and Sakurai *et al.* [5] with flat-ribbon heaters, the predicted values were much smaller than the measured values.

Figures 6–9 show the comparison of Eq. (16) with the data of Sakurai and Shiotsu [8] at various pressures. In these experiments, Sakurai and Shiotsu [8] used a horizontal wire of 1.2 mm in diameter. The power was increased exponentially according to Eq. (18). The comparisons are quite successful for high pressures with the data slightly overpredicted for lower values of τ. The following factors must be considered to improve the predictive capability of the present theory:

1. The quasi-steady conduction model may not be accurate especially for very small values of the exponential period, τ. In such cases, better coupling is necessary between the conduction and convection fields.
2. The idealized bubble assumption may yield some error in predicting the hovering period. Even a small error may be very important in boiling with very small values of τ.

As shown in Figs. 6 and 7, for pressures of 0.196 and 0.101 MPa, the present theory overpredicts the data for values less than 50 ms. This is expected, since Sakurai and Shiotsu [8] observed that in these experiments at low pressure and small exponential period boiling pattern of *type-II* is present. In Figs. 6–9, the dotted line represents the case in which the liquid layer thinning is due to evaporation alone, as suggested by Serizawa [9]. As shown in these figures, for small values of τ this thinning model yields considerable error.

Figure 10 shows the comparison of the present correlation with the data of Kuroda (as cited by Serizawa [9]). The results are the same as the results of the comparison with the data of Sakurai and Shiotsu [8] since the same experimental setup was used. In the same figure, the present theory is also plotted using a value for the hovering period 10% less than the predicted value from Appendix A. This illustrates the effect of the accuracy in predicting the hovering period.

The present theory is also compared to the data of Tachibana *et al.* [4] as shown in Fig. 11. In this experiment, flat-ribbon heaters were used. The heat generation rate per unit surface area was increased linearly. Thus, assuming that the quasi-steady conduction model is valid, Eq. (13) is used to evaluate η. However, the experimental observations of Tachibana *et al.* [4] show that the surface heat flux does not increase linearly as the heat generation rate per unit surface area does and that dq/dt is greater than dQ_g/dt after the initiation of boiling. The experimental study of Kawamura *et al.* [6] was very similar to the experiment of Tachibana *et al.* [7] Thus, the results as compared to the present theory are the same.

The data of Sakurai *et al.* [5] are also compared to the present correlation, as shown in Fig. 12. In this experiment, flat ribbon heaters of very small geometry (0.1 × 2.5 × 15 mm) are used. Since the unstable Taylor wavelength is larger than the longest heater dimension, the geometry effects become dominant and the hydrodynamics become very complex. The

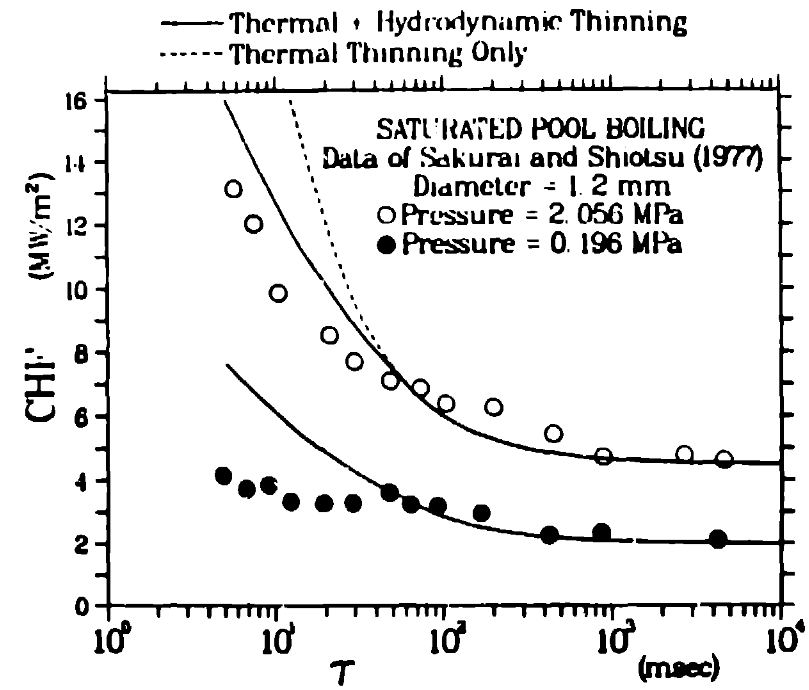

Fig. 6.

Comparison of the present theory with the data of Sakurai and Shiotsu with 2.056 and 0.196 MPa pressure [8].

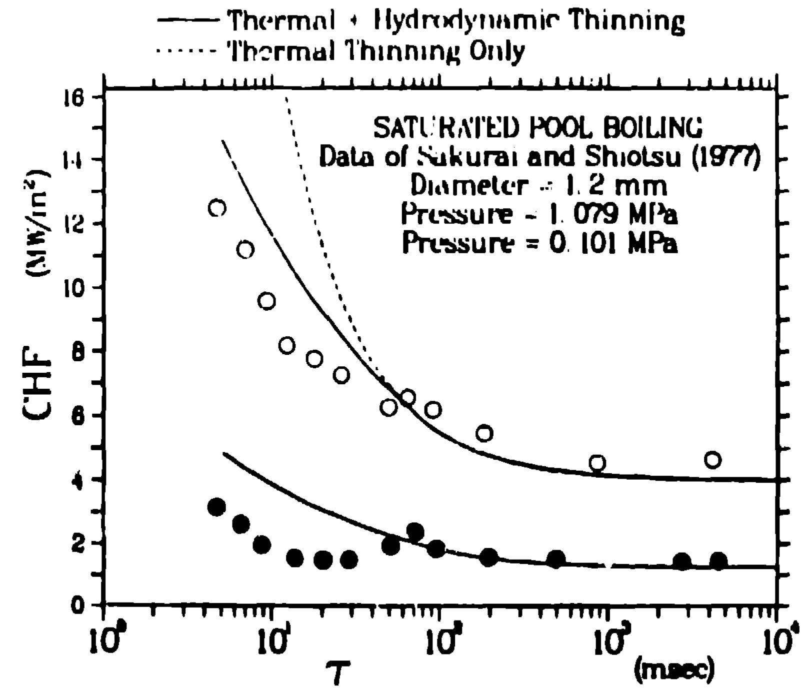

Fig. 7.

Comparison of the present theory with the data of Sakurai and Shiotsu with 1.079 and 0.101 MPa pressure [8].

experiments were simulating exponential power increase: therefore, Eq. (16) is used to calculate η. As shown in Fig. 12, the present theory is only good in predicting the general trend of the data. Various factors caused by the flat-ribbon heater geometry that affect the results of the present theory may be summarized as follows:

1. The hydrodynamics of boiling with flat-ribbon heaters is not well known. An equivalent diameter approach described in Appendix A for predicting the bubble-hovering period may not be accurate. The same approach yields poor prediction of even the steady-state CHF as reported by Haramura and Katto [11].
2. Flat-ribbon heaters used in the above experiments have the broad heat-transfer surface oriented vertically. In such geometry, especially for very small heaters, an assumption of no liquid supply during the hovering period may not be very realistic. Also, for horizontal cylindrical heaters of small length, the liquid supply mechanism may be important.
3. It is also interesting to note that in both experiments [4, 5], the measured values of the steady-state CHF were about 50% lower than the values predicted by using the available correlations [11, 18]. It is likely that the experimental results are altered by the end effects.

The experimental setup of Kataoka *et al.* [9] was designed to study the transient CHF during forced convective boiling. In this setup, which consists of a vertical wire placed in a vertical flow channel, Kataoka *et al.* [9] also obtained a set of data with zero flow and zero subcooling. The comparison of the present theory to this set of data is shown in Fig. 13. The applicability of the present theory to this specific situation is questionable because of the vertical heater geometry. The use of the Taylor instability wavelength and the estimation of the hovering period based upon an idealized bubble equation of motion may be very approximate with this geometry. However, Kataoka *et al.* [9] were able to correlate their zero-flow-zero subcooling CHF data using Zuber's [12] saturated pool boiling correlation with Kutadelatze's [19] constant. This suggests that the boiling configuration with a vertical heater may be very similar to the configuration with horizontal heater. The present theory overpredicts the data at 0.143 MPa. This is because of *type II* boiling present at this low pressure. At 0.594 MPa, the comparison between the present theory and the data is exceptionally good.

VI. SUMMARY AND CONCLUSIONS

In this study, a theoretical prediction of the CHF during power transients in saturated pool boiling is presented. A general transient CHF model is developed. In this model, the liquid layer thinning at heat fluxes in excess of the steady-state CHF is formulated as caused by either the hydrodynamic or the thermal mechanism. The switch-over between the two mechanisms is graphically illustrated. A represantative mechanism map for exponential transients is also shown in the paper.

Based upon this general model, closed-form correlations to predict the transient CHF in exponential and linear transients are obtained. The effect of the conduction within the heater on the present theory is also included. Using the quasi steady conduction model, the theory is compared with the data quoted in the international literature. It is observed that the comparison for horizontal cylindrical heaters at high pressures is quite favorable. At lower pressure, the discrepancy is thought to arise because of the inadequacy of the quasi steady conduction assumption. Although successful in predicting the general trend, the theory

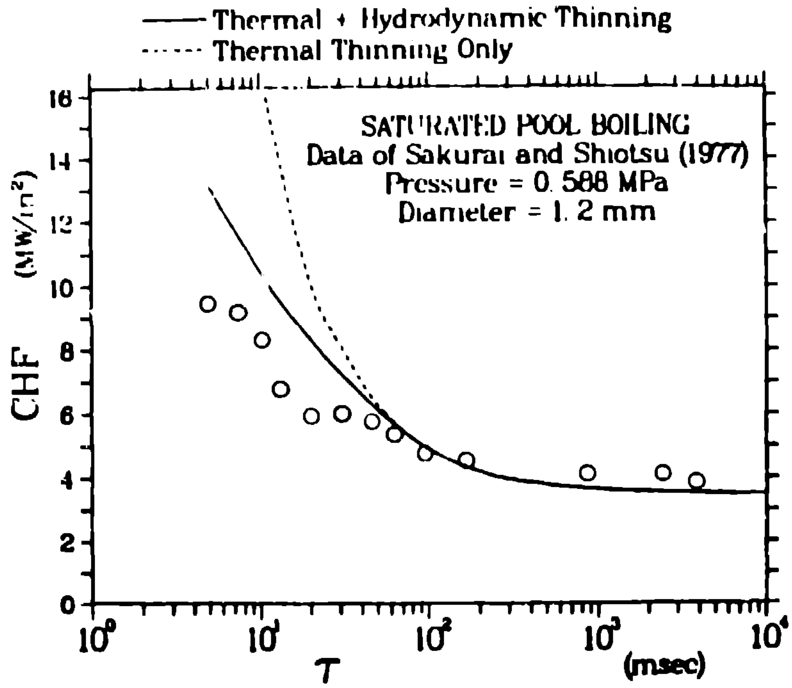

Fig. 8.

Comparison of the present theory with the data of Sakurai and Shiotsu with 0.588 MPa pressure [8].

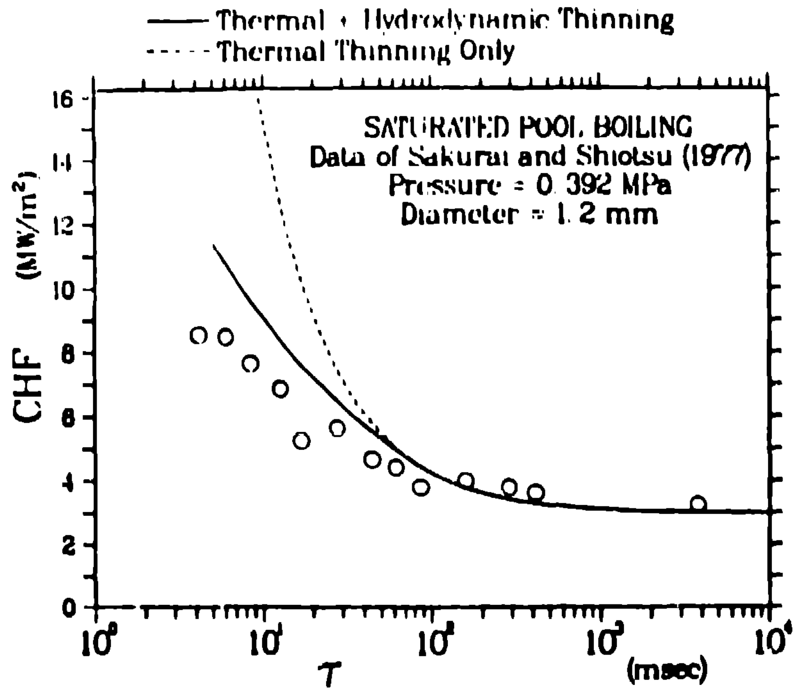

Fig. 9.

Comparison of the present theory with the data of Sakurai and Shiotsu with 0.392 MPa pressure [8].

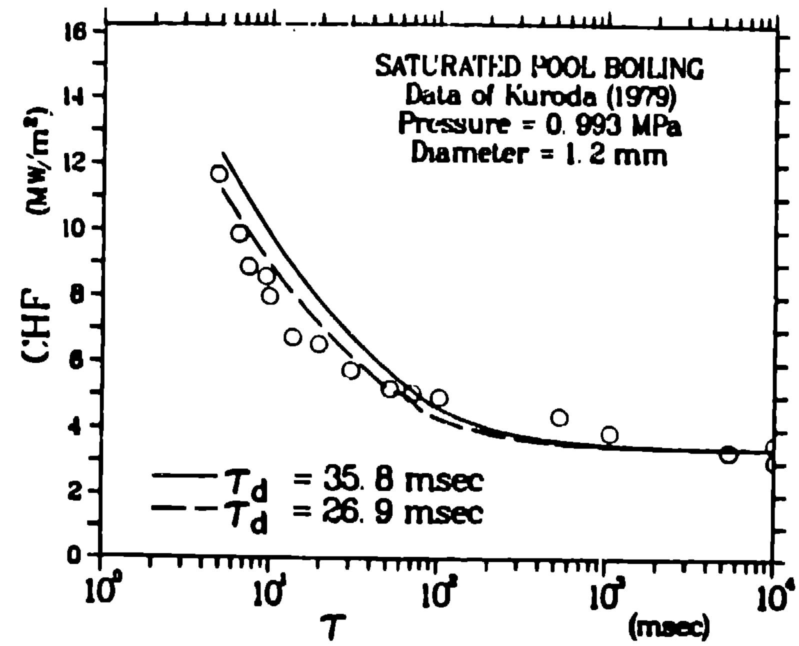

Fig. 10.
Comparison of the present theory with the data of Kuroda (as cited by Serizawa [9]).

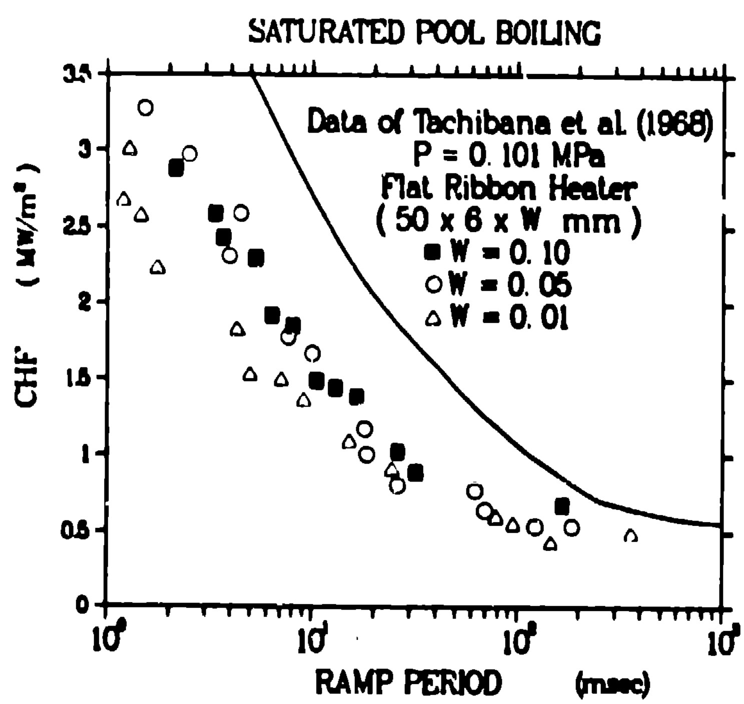

Fig. 11.
Comparison of the present theory with the data of Tachibana *et al.* [4].

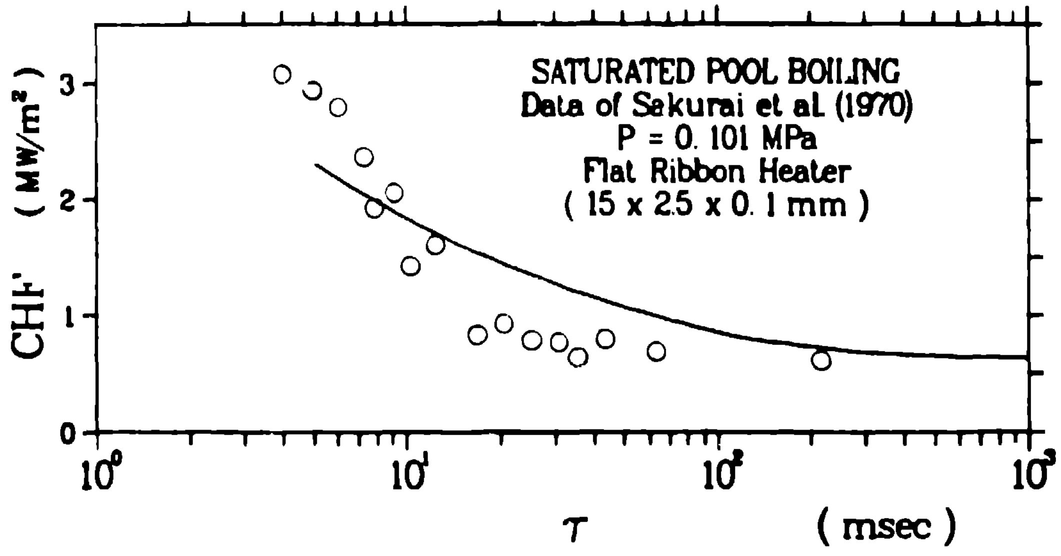

Fig. 12.
Comparison of the present theory with the data of Sakurai *et al.* [5].

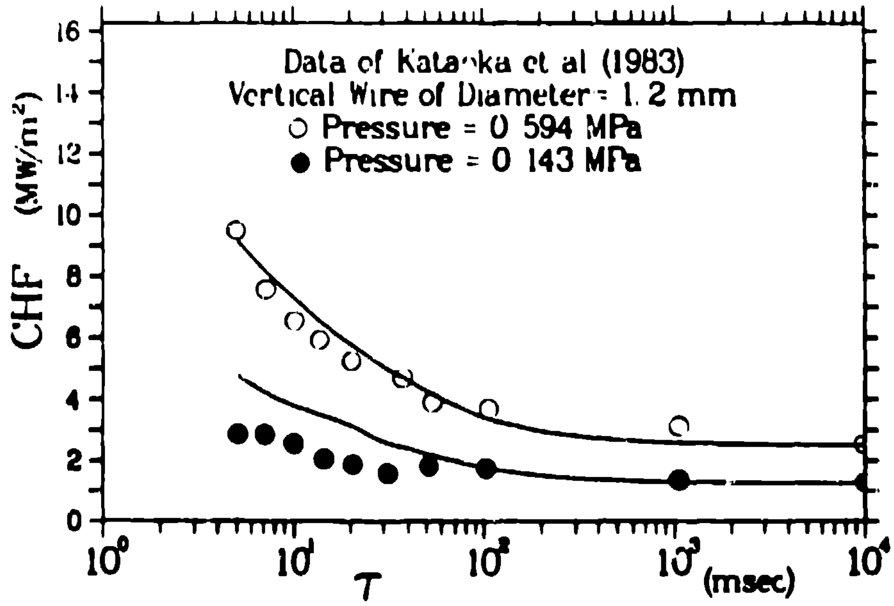

Fig. 13.
Comparison of the present theory with the data of Kataoka *et al.* [10].

resulted in a higher discrepancy when compared to CHF data from flat ribbon heaters. Various reasons for this discrepancy are summarized. Among those, the most important is thought to be the unknown hydrodynamics of flat-ribbon heaters.

In this paper, for the first time in the open literature, the effect of hydrodynamic thinning mechanism on the transient CHF is discussed. Its importance is shown through data comparison. An improvement in the predictive capabilities of the present theory is possible if (1) the hydrodynamics of flat-ribbon heaters are better known, (2) a better coupling between the heat-generation rate and the surface heat flux is established, and (3) the bubble-hovering period is better estimated.

APPENDIX A

BUBBLE HOVERING PERIOD IN SATURATED POOL BOILING

In this appendix, the equation given by Haramura and Katto [11] to estimate the hovering period is summarized. This equation is obtained by solving the equation of motion of an idealized bubble [20]. The summary includes the cylindrical heater geometry only. Haramura and Katto [11] formulated the hovering period in the following form:

$$\tau_d = (3\pi/4)^{0.2}(4/g)^{0.6}[(\xi\rho_f + \rho_g)/(\rho_f - \rho_g)]^{0.6} v_1^{0.2} \quad , \tag{A1}$$

where v_1 is the volumetric growth rate of the bubble and is the volumetric ratio of the accompanying liquid to the moving bubble. The term ξ is taken as $\frac{11}{16}$ by Haramura and Katto.

For cylindrical heaters, the heater area contributing to one vapor jet is $\pi d\lambda'_D$, and the volumetric grow rate of the bubble is

$$v_1 = \pi d\lambda'_D q/(\rho_g h_{fg}) \quad . \tag{A2}$$

where λ'_D is the modified unstable Taylor wavelength. It is modified to include the additional effect of the surface tension along the curvature, and is given by

$$\lambda'_D = \frac{2\pi\sqrt{3}\,[\sigma/g(\rho_f - \rho_g)]^{\frac{1}{2}}}{[1 + 2\sigma/d^2 g(\rho_f - \rho_g)]^{\frac{1}{2}}} \quad . \tag{A3}$$

In the present study, the hovering period needs to be estimated at steady-state CHF level. Thus, q in Eq. (A2) must be replaced by $q_{CHF,SS,OO}$.

For flat-ribbon heaters, a simple approach to estimate the hovering period is suggested by Haramura and Katto [11], where the above procedure is followed after replacing the heater diameter by an equivalent diameter, d_e, given by

$$d_e = \frac{2W}{\pi} \quad . \tag{A4}$$

APPENDIX B

THE EFFECT OF VAPOR ACCELERATION ON THE LIQUID LAYER THICKNESS

Assuming that the quasi-steady approach may be used to determine the Helmholtz instability wavelength, λ_H, the theoretical wave velocity equation at the vapor/liquid interface may be written as

$$c = -\left[\frac{1}{(\rho_f + \rho_g)}\frac{2\pi}{\sigma} - \frac{\rho_f \rho_g}{(\rho_f + rho_g)^2}(u_g + u_f)^2\right]^{\frac{1}{2}} . \qquad (B1)$$

Using the continuity and heat-balance eq ıtions, it can be shown [11] that the liquid velocity, u_f, is much smaller than the vapor veloc , u_v. Thus, solving for the wavelength, λ, which corresponds to a zero wave velocity, c, the Helmholtz instability wavelength may be obtained as

$$\lambda_H = 2\pi\sigma\frac{\rho_f + \rho_g}{\rho_f \rho_g}\frac{1}{u_g^2} . \qquad (B2)$$

However, when the surface heat flux is a function of time, the vapor velocity becomes time-dependent. It is related to the surface heat flux by means of the energy balance as follows:

$$u_g = \frac{A_w}{A_v}\frac{q}{\rho_g h_{fg}} . \qquad (B3)$$

Therefore, under transient heat-flux conditions, the inlet vapor velocity at the vapor stems is different from the exit vapor velocity. As long as the liquid layer is thin, the exit velocity may be expressed in terms of the inlet velocity as follows:

$$u_g(t + \delta t) = u_g(t) + \frac{du_g}{dt}\frac{\delta}{u_g(t)} . \qquad (B4)$$

where $u_g(t + \delta t)$ is the inlet velocity, $u_g(t)$ is the exit velocity, and δ is the liquid-layer thickness. Eq. (B4) is correct for a constant acceleration (for linearly increasing surface heat flux). For time-dependent acceleration (for exponentially increasing surface heat flux, for instance), Eq. (B4) represents the Taylor series approximation for small δ/u_g. For an exponential transient, Eq. (B4) may be written as

$$u_g(t + \delta t) = u_g(t) + \frac{\delta}{\tau} . \qquad (B5)$$

where τ is the exponential period. Haramura and Katto [11] assumed that the maximum stable magnitude of the liquid layer thickness is a quarter of the Helmholtz instability wavelength as given by Eq. (1) in the text. In the present study, it is assumed that the system responds very quickly to the transient. Therefore, the vapor velocity at the inlet of the vapor stems is used to evaluate the Helmholtz wavelength, which is referred to as $\lambda_{H,min}$. On the other hand, for a system responding very slowly, the exit velocity, $u_g(t)$, must be used to evaluate

λ_H. This determines the upper limit of λ_H and is called $\lambda_{H,max}$. Using Eqs. (B2) and (B3), the following relation between $\lambda_{H,min}$ and $\lambda_{H,max}$ may be obtained:

$$\lambda_{H,min} = \lambda_{H,max}\left(1 + \frac{\lambda_{H,max}}{4u_g(t)\tau}\right)^{-2} \approx \lambda_{H,max}\left(1 - \frac{\lambda_{H,max}}{4u_g(t)\tau}\right)^{2} . \tag{B6}$$

Equation (B6) suggests that the difference between $\lambda_{H,min}$ and $\lambda_{H,max}$ becomes important as the term $\tau u_g(t)$ becomes small. The minimum value of u_g used in the present study is the one corresponding to the steady-state CHF at atmospheric pressure. The minimum magnitude of practical interest for τ is 5 ms. Even for these extreme cases, the solution of Eq. (B6) yields a difference of less than 1% between the upper and lower limits of λ_H. Therefore, it is expected that estimating the liquid-layer thickness through a quasi-steady approach must be reasonably accurate.

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